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Refrigerant Migration in a Split-Unit Air Conditioner

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ABSTRACT

The relationship between cyclic refrigerant migration and cyclic loss for a residential, split-system air conditioner has been investigated. The cyclic refrigerant migration was measured at different points in the operating cycle by simultaneously shutting five pneumatically operated valves that isolated the refrigerant in the major system components. The refrigerant was then removed, weighed, and returned to the system. The unit tested was found to have a high initial capacity as migrated refrigerant was removed from the evaporator and then a low, slowly increasing capacity as trapped refrigerant was returned to the system from the accumulator. The unit performance was also compared to single and double time constant regressive approximations and to the time constant calculated from the evaporator mass and heat transfer coefficient. Although relationships between migrated refrigerant and cyclic capacity were observed, no practical refrigerant migration test method that would be less burdensome than the cyclic tests of ASHRAE Standard 116 appears possible at this time.

INTRODUCTION

Because of the concerns about test cost and testing time requirements of the current evaluative procedure for central air conditioners, the Department of Energy (DoE) requested the National Bureau of Standards (NBS) to investigate several alternative approaches for determining cyclic degradation that might result in a simpler, cheaper test procedure without sacrifice of repeatability or of comparability to results obtained currently.

Laboratory and field tests have shown that units that do not allow off-cycle refrigerant migration (i.e., units with thermostatic expansion valves without bleed ports) have less performance degradation when cycling than do units that allow off-cycle pressure equalization and consequent refrigerant migration. As a result, the existing cyclic test procedures were conceived to allow for credit to be given for those designs that might minimize cyclic losses. Modeling efforts that might be used to mitigate some of the testing requirements were thwarted, since the amount of migration varies from one design to another and is also a function of operating conditions. Therefore, it was not possible to characterize an air conditioner by a simple time constant, as was done for the furnace/boiler evaluative procedure. Theoretical approaches to cyclic performance analysis have been presented which use refrigerant migration during the off-cycle as an input (Chi and Didion 1982; Nguyen, et al., 1981). However, no adequate empirical data has existed in the public domain to quantify the refrigerant flow during the dynamic periods of a typical air-conditioner operation.

This report presents results of dry coil steady-state and comparable cyclic tests performed at the DOE/NBS test procedure (Kelly and Parken 1978) rating point (82 F or 27.8°C outdoor temperature) using the recently introduced continuous indoor fan option for the cyclic tests (ASHRAE Standard 116). The refrigerant mass in five major areas (compressor and accumulator, liquid line, vapor line, condenser, evaporator) was determined by shutting quick closing,

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pneumatic ball valves, isolating these areas and then extracting and weighing the refrigerant contained in each. Refrigerant mass distribution data were taken for steady-state operation and at the beginning, end (six minutes) and at several intermediate times during an on cycle performed as an extension of six minutes "on" and 24 minutes "off". Cyclic tests were performed in accordance with the DOE/NBS test procedure (Kelly and Parken 1978).

No adequately confirmed theoretical model for calculating the cyclic degradation factor from this experimental data or recommended alternate test procedure is presented at this time. This empirical study indicates that such a procedure may be possible; although the representative function will be more than a simple time constant, and the method for obtaining the test data may not be less burdensome than the current test methods.

The test apparatus and procedure used in this study are described in detail in Mulroy and Didion (1983).

TEST SPECIMEN AND FACILITY

A nominal three ton, split system, air-to-air unitary heat pump operating in the cooling mode was used for this investigation. The unit, being a heat pump, differed from a typical split system air conditioner in its coil sizing and in possessing a four way valve and an accumulator. These design changes generally result in a heat pump having a lower steady-state efficiency and a greater cyclic degradation coefficient than that of a central air conditioner of similar cooling capacity. It was believed that this expected increase in the variable to be studied, the cyclic degradation factor, would increase the measuring accuracy. The unit was equipped with a crankcase heater automatically energized during all off-cycles. The outdoor unit contained the compressor, outdoor coil, fan defrost controller, and associated equipment. The indoor and outdoor sections of the heat pump were installed in separate but adjoining environmental chambers.

System measurements not generally considered in a performance test, but of particular interest in this study were the volumes and masses of various system components, some of which are summarized in Table 1 and 2. The volume measurements in Table 1 listed as experimental were determined by vapor charging the system component volumes isolated by pneumatically operated ball valves (Figure 1) with refrigerant 22. The system volumes were then calculated from the specific volumes of the thermally stabilized refrigerant in each section (taken from the R22 superheat tables at the measured temperatures and pressures) and the measured refrigerant mass introduced by taking before- and after-charging cylinder weights (correcting for charging hose volume). As a check, comparable volumes listed as dimensional in Table 1 were calculated from tubing lengths, outside diameters, and presumed wall thicknesses for all components except the compressor-accumulator section, which was too complex to allow a reasonably accurate volume estimate from the available dimensional data.

The component mass measurements given in Table 2 were calculated using the dimensional data used in preparing Table 1 with the addition of measurements of the coil fin dimensions.

The unit was tested in the environmental chambers and apparatus described in Kelly and Parken (1978). The primary method used to measure the cooling capacity was the indoor air enthalpy method. The secondary method used was the volatile refrigerant flow method. Both of these methods are described in ASHRAE Standard 37.

In addition to the test setup described in Kelly and Parken (1978), pneumatically operated ball valves operated by a solenoid pilot were installed in the locations shown schematically in Figure 1, so that portions of the system could be isolated and their refrigerant content determined by transferring the refrigerant to a canister and determining the change in refrigerant container weight.

TEST PROCEDURE

Prior to each day's testing, the unit was vapor charged with the same amount of refrigerant that was removed after the previous day's test.

It was found that the unit performance for equally calculated total charges would be different if charged from a vacuum as opposed to charging from a residual vapor charge. In this case the unit charged from an evacuated state would be undercharged because of refrigerant absorption by the crankcase oil. For all tests in this report the unit was charged from a residual vapor charge that would result in a quantity of refrigerant absorbed in the oil near the steady-state value.

When steady-state operation of the unit and test chamber reached the 82 F (27.8°C) dry coil test condition of the DOE/NBS test procedure, data were taken for a minimum period of 1/2 hour. For the three tests reported as QC 3, QC 5, and QC 6 in Table 3, the unit was shut off and the pneumatically operated valves simultaneously closed at the end of the steady-state data period. For the other reported tests, manually-operated cyclic operation was begun at the end of the steady-state data period. These cyclic tests were performed following the guidelines of the DOE/NBS test procedure using the six minute on cycle length and 24 minute off cycle length. All tests were performed with continuous indoor fan operation (i.e., during both the on- and off-cycles).

Three complete cycles were performed to allow conditions to become repeatable in the cyclic mode. A fourth cycle was then performed during which DOE/NBS test procedure test data were taken. Following this "data" cycle an additional 24-minute "off" period followed by a six minute "on" period were performed. The pneumatically operated valves were closed at various times between cut-on and six minutes of operation during this "on" period. The compressor was wired to stop simultaneously with pneumatically operated valve closure.

In all cases, closure of the pneumatically operated valves was immediately followed by extraction and weighing of the refrigerant in each isolated section.

The refrigerant content of each isolated section, for the pneumatically operated valve tests, was calculated as the weight removed plus the mass of the vapor remaining calculated from the section volume and the density of superheated vapor at the final pressure and temperature recorded after sufficient time had been allowed for these quantities to stabilize. The quality (mass fraction of vapor) in each isolated section was calculated from the above determined total refrigerant mass and the refrigerant vapor and liquid densities calculated from the pressures and temperatures occurring at the time of valve closure. In this case, the vapor was assumed to be superheated at the average temperature of each section, and the liquid was assumed to be saturated at the lowest temperature measured by any of the component thermocouples on that section or, if lower, the section saturation temperature corresponding to the measured pressure. The oil contant of the refrigerant samples was not analyzed however close agreement between the amount of refrigerant charged to the system and the amount removed in sum from the isolated sections indicated that the cyclic oil migration was not significant at this operating condition.

Only one test, QC 11, was performed with the unit's indoor fan cycling with the compressor. This test was terminated at the end of a 24 minute cut-off period with pneumatic isolation valve closure at that time. As can be seen in Table 3, the off-cycle refrigerant migration for this cyclic fan test was not significantly different from the continuous fan test data.

DISCUSSION AND RESULTS

The refrigerant migration data taken with the quick-closing pneumatically operated valve system are listed in Tab. 3 and 4 and are plotted as a function of time in Figure 2. Some commments that can be made about this data are:

1. Most of the refrigerant was on the high-pressure side during steady-state operation. Approximately 46% of the total charge was in the condenser and 37% in the liquid line for a total of 83%, the remaining 17% being on the low-pressure side.

2. Most of the refrigerant was on the low-pressure side at the end of the off-cycle. Approximately 56% of the total charge was in the evaporator, 2% in the vapor line, and 22% in the compressor and accumulator for a total of 80%, the remaining 20% being on the high pressure side.

Refrigerant migration to the evaporator during the off-cycle was initially the result of the pressure gradient between the condenser and evaporator. After these pressures equalized, which was typically in about five minutes for this unit, migration continued as a result of the temperature gradient between the condenser and evaporator. The refrigerant migration to the accumulator during the off-cycle was presumably by gravity flow from the evaporator, since the accumulator in the compressor compartment was considerably warmer than the evaporator, during the off-cycle. At the end of a typical off-cycle, temperatures of 94.1 F (34.5°C) and 82.6 F (28.1°C) were recorded from insulated thermocouples soldered to the top and bottom of the accumulator, respectively. This bottom temperature was consistent with the presence of saturated liquid and the top temperature was indicative of the high temperature in the compressor compartment, indicating that this liquid refrigerant could not have been transferred to the accumulator by thermal gradients.

Some liquid refrigerant was also found in the suction line at the end of the off-cycle. It is presumed that during the off-cycle, liquid refrigerant flowed by gravity from the evaporator through the suction line to the accumulator and that refrigerant vapor was transferred in the opposite direction by the existing temperature gradient. The indoor unit was installed in the test setup at an elevation 13 inches (33 cm) higher than the outdoor unit to accomodate its bottom air duct.

There was no evidence of substantial liquid refrigerant transfer to the compressor crankcase during the off-cycle or on start-up. During a typical cyclic test, an insulated thermocouple, soldered to the bottom corner of the compressor shell indicated a steady temperature rise from 149 F (65.0°C) at cut-on to a temperature of 169 F (76.1°C) after six minutes of operation. A sharp decrease in the temperature would have occurred had there been a significant quantity of liquid refrigerant in the compressor crankcase. Consequently, it was assumed that all the refrigerant in the liquid phase in the compressor-accumulator section was in the accumulator.

3. Most of the refrigerant that was in the evaporator at the end of the off cycle left within 15 seconds after start-up. The amount of refrigerant in the evaporator reached a minimum at approximately 1.5 minutes after start-up and then steadily increased to its steady-state value.

4. The amount of refrigerant in both the condenser and liquid line increased steadily from their start-up to their steady-state values.

5. The amount of refrigerant in the compressor and accumulator section reached its maximum recorded amount of refrigerant (38% of the total charge) at the data point 15 seconds after start-up. The amount of refrigerant in this section slowly decreased to its steady-state value (5% of the total charge) as the accumulator gradually returned its liquid refrigerant to circulation. Clearly much of the cyclic loss can be attributed to the unit being in effect "undercharged" while the liquid refrigerant is stored in the accumulator. This point is emphasized in Figure 3 in which the unit capacity and the temperature measured by a thermocouple soldered to the bottom corner of the accumulator minus the compressor is saturated suction temperature are plotted against time. Here it is seen that the capacity does not reach its steady-state value until the accumulator's bottom temperature departs substantially from the compressor's saturated suction temperature, an indication that the accumulator has been emptied of liquid refrigerant.

Further comparison of the refrigerant migration data to unit performance requires calculation of the refrigerant side capacity. Figure 4 shows the measured air-side capacity for the first six minutes of operation and a refrigerant side capacity calculated from this air-side data.

The refrigerant side capacity was calculated by drawing a smooth curve through the air side data points and then assuming that, for small time intervals (0.1 minute),the air side capacity was exponentially approaching a constant refrigerant side value. Expressed as an equation this would be:

$$\dot{Q}_{REF} = \dot{Q}_{AIR}(t_1) + \frac{\dot{Q}_{AIR}(t_2) - \dot{Q}_{AIR}(t_1)}{1 - \exp\left(\frac{t_1 - t_2}{\tau}\right)}$$

<u>where</u> $\dot{Q}_{REF}$ = refrigerant-side capacity

$\dot{Q}_{AIR}$ = air-side capacity

t_1, t_2 = times 1 and 2, min

τ = evaporator time constant, MC/UA, min

The evaporator time constant was calculated from the coil mass data given in Table 2 and from the steady-state test value of UA (equal to the steady-state capacity divided by the air-to-refrigerant saturation temperature difference), giving a value of 0.63 minutes for the evaporator time constant. In Figure 5 this calculated refrigerant side capacity is compared to the mass of refrigerant in the evaporator. The two curves follow one another in shape quite well after 15 seconds and lead to the following interpretation for the shape of the refrigerant-side capacity curve.

These results may be explained by considering that at cut-on, there is a sufficient amount of refrigerant in the evaporator to allow near steady-state capacity, even though the capillary tube is not feeding sufficiently to sustain this capacity level. As the refrigerant leaves the evaporator by nucleate boiling and slug flow, liquid is carried into the accumulator. This drops capacity to a level that can be sustained by the flow that the capillary will permit. This "capillary-limiting" capacity value is reduced below the steady-state value primarily because the unit is effectively "undercharged" by the amount of liquid temporarily lying in the accumulator. As the accumulator returns its retained liquid to the system, the condenser pressure increases, causing the capillary flow rate to increase toward its steady-state value. This phenomena is further complicated by the lack of a stable liquid seal ahead of the capillary during this period.

The unit sight glass was observed during several cycles and found to be 1/2 full after 30 seconds of operation. The outlet port of the sight glass was 3/4 full after 1 minute of operation. After 1 minute and 45 seconds, the outlet port was covered, but the bubble in the top of the sight glass occaisionally connected to the outlet port until 4.5 minutes after start-up. This indicated that total liquid stability between the condenser outlet and the capillary tube entrance had not yet been established. The sight glass was completely clear 5.5 minutes after start-up. The pseudo-high refrigerant capacity noted in the first minute then is a result of the excess of evaporator refrigerant accumulated during the off-cycle and, as this is "pumped" out," the capillary feed tends to starve the evaporator until, gradually, a steady flow rate is achieved.

The two sets of test results, shown in Figure 6, were obtained by allowing the unit to come to steady-state capacity after several 6 minute-on/24 minute-off cycles. They differed in the indoor air dew point, which was 2.8°C (37.0 F) for test QC 26 and -6.4°C (20.5 F) for test QC 27. The second test (at the lower dew point) was performed to determine if, when a very low saturation temperature occurred during the starved evaporator period, there would be a significant intracycle latent cooling effect. It was concluded from the closeness of these two curves that this was not a significant concern.

The insight provided by Figure 5 is useful in preparing a regressive fit to the capacity as a function of time as presented in Figure 6. The experimental data shown in Figure 6 were fitted with two curves. In the first, a single time constant exponential curve, $Q/Q_{ss} = (1 - e^{-t/\tau 1})$, where $\tau 1$ = 3.0 minutes, satisifies the 0 and infinity time conditions and matches the transient data values fairly well after four minutes of operation. However, before that time the indicated capacity is much too low.

If a second time constant in the form of a term $(1 + a\, e^{-t/\tau 2})$, where a = 4.0, $\tau 2$ = 0.8 minutes, is introduced to simulate the very high, but rapidly decaying, initial capacity shown in Figure 5, the resulting equation is

$$\dot{Q}/\dot{Q}_{ss} = (1 - e^{-t/\tau 1})(1 + a\, e^{-t/\tau 2}) \tag{5.2}$$

This equation form, when fitted to the data, provides a very good trace of the phenomena observed. It should be noted that the terms $\tau 1$ and $\tau 2$ in these equations have now become regressive curve fitting constants and are not time constants in the classical transient heat-transfer capacitance sense of Equation (5.1).

An additional test was performed to record the capacity decay after shut-down. The results of this test are shown in Figure 7. The regressive least-squares fit to this data resulted in the shown time constant of 0.61 minutes. During this off-cycle, it is not surprising that a simple decay function should simulate the transient capacity so well, since none of the accumulator or capillary complications are significant. Furthermore, it is interesting to note that for this air-conditioning unit, a similar coefficient value may be construed for the start-up period.

As shown in Figure 4, the unit refrigerant-side capacity for the first 3/4 minute of operation is near the steady-state value. Making the assumption of constant refrigerant-side capacity for the first 3/4 minute of operation, a regressive exponential least-squares fit of the form $y = ae^{bx}$ was made to the experimental air-side data by rearranging the unit's air-side capacity to the form:

$$1 - \dot{Q}/\dot{Q}_{ss} = ae^{bx}, \text{ where } b = -1/\tau$$

The results of this fit were a = 0.99 and τ = 0.80 minutes. The value of τ is reasonably close to the value of 0.63 minutes calculated from the coil mass, specific heat, and heat transfer rate. A somewhat higher experimental time constant could be attributed to additional system masses in addition to the evaporator and to the measurement system. This time constant being equal to $\tau 2$ in Equation 5.2 is purely a coincidence.

CONCLUSIONS

The empirical results of this study suggest that it is possible to simulate the dynamic operation of an air conditioner on two levels of complexity. The first level, a single time constant characterizing the start-up, and shut down operation, tends to ignore some of the actual physical phenomena that occur; however, as an approximation or a next step past steady state toward reality, this may be adequate. Such an approach may be considered when the model is used for characterizing a "typical" or the "general class" of air conditioners. Unfortunately, a time constant value that is the best representative of the state of the art is not possible to determine without considerable further study of many different units. The value obtained for this unit is undoubtedly a typical value, but how wide the range or what the median is for the air-conditioning units of today is not known.

The second level of simulation is that which is of the form shown in Equation 5.2. Here both facets of the cyclic degradation are accounted for, and such a form could be used to characterize the operation of any given air conditioner. This level of simulation and its respective empirical data is essential if the intention of the simulation is to distinguish among individual units' characteristics or dynamic performance; that is to say, if proper credit is to be attributed to a specific design for rating purposes. If the air-side capacity of a specific unit were to be determined, as shown in Fig. 4, from a continuous plot of the air-side thermopile temperature difference, that air conditioner's dynamic performance would probably be simulated as accurately as is done by the current cyclic tests. However, between the premeasurement operations required to assure periodic repeatability and the confirming shut-down dynamic measurements, the experimental burden is no less than the existing cyclic tests. It would therefore appear that the best hope for a less burdensome characterization of a given unit's dynamic performance is through sufficient study of the individual components' interaction and impact on refrigerant migration and then an establishment of a series of enhancement (positive or negative) factors, which could be attributed to a given unit's steady-state performance depending on its particular design.

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APPENDIX A

Comparison of Cyclic Performance to Reduced Charge Steady-State Performance

In an attempt to verify the "starved evaporator" concept suggested, several steady-state tests were performed with the refrigerant charge reduced for comparison to the transient data taken during the period in which liquid, trapped in the accumulator, was believed to be the major cause of cyclic loss. These data are shown in Figure A1 and capacities calculated from the indicated linear least squares fit to the capacity data in Figure A1 are compared to the transient capacity from Figure 4 in Figure A2. The capacity from 1.5 minutes after start-up tp six minutes after start-up exceeds the value calculated from the transient data by approximately 10%. The air-side capacity, which is equivalent to the refrigerant side capacity calculated from the accumulator contents and reduced charge steady-state capacity, as shown in Figure A2, is compared to the experimental air-side capacity in Figure A3.

TABLE 1

Component Volume Measurements

	Dimensional	Experimental
Outdoor Coil	263 in^3 (4310 mL)	306 in^3 (5020 mL)
Compressor and Accumulator	--	610 in^3 (10000 mL)
Vapor Line	61 in^3 (1000 mL)	53 in^3 (870 mL)
Liquid Line	63 in^3 (1030 mL)	78 in^3 (1280 mL)
Indoor Coil	314 in^3 (5150 mL)	346 in^3 (5760 mL)

TABLE 2

Component Mass Measurements

Outdoor Coil	27.4 pounds (12.4 kg) of copper 25.1 pounds (11.4 kg) of aluminum
Indoor Coil	36.7 pounds (16.7 kg) of copper 19.6 pounds (8.9 kg) of aluminum
Vapor Line	6.5 pounds (3.0 kg) of copper

TABLE 3

Refrigerant Content of Isolated Sections

TEST	DESCRIPTION	LIQUID LINE		VAPOR LINE		INDOOR COIL		OUTDOOR COIL		COMPRESSOR & ACCUMULATOR		TOTAL
		kg	%	kg	%	kg	%	kg	%	kg	%	kg
QC #7	Off, 24 min	.35	8.7	.09	2.3	2.23	56.1	.41	10.4	.89	22.5	3.97
QC #8	Off, 24 min	.36	9.1	.09	2.3	2.21	56.0	.44	11.3	.84	21.4	3.95
QC#11	Off, 24 min*	.22	5.8	.03	1.3	2.31	60.0	.34	8.7	.93	24.2	3.85
QC#20	On, .25 min	.83	21.0	.03	1.3	.33	8.9	1.23	31.3	1.48	37.6	3.94
QC#21	On, .50 min	.92	23.1	.03	0.7	.32	8.1	1.37	34.5	1.33	33.6	3.96
QC#16	On, 1.00 min	1.09	27.9	.02	0.5	.18	4.7	1.53	39.4	1.07	27.5	3.89
QC#17	On, 1.50 min	1.18	30.5	.02	0.5	.16	4.2	1.51	38.9	.99	25.5	3.88
QC#18	On, 2.00 min	1.26	32.1	.02	0.5	.22	5.7	1.49	38.1	.93	23.7	3.91
QC#19	On, 4.00 min	1.39	35.5	.02	0.5	.28	7.1	1.54	39.4	.69	17.6	3.92
QC #9	On, 6 min	1.43	36.5	.02	0.5	.32	8.2	1.65	42.1	.49	12.6	3.93
QC#10	On, 6 min	1.43	36.6	.02	0.5	.32	8.3	1.66	42.6	.46	11.9	3.89
QC #3	Steady-state	1.43	37.0	.03	0.7	.40	10.5	1.78	46.1	.22	5.8	3.85
QC #5	Steady-state	1.43	36.5	.02	0.6	.43	10.8	1.84	46.7	.21	5.4	3.95
QC #6	Steady-state	1.43	36.4	.02	0.6	.43	10.9	1.84	46.9	.21	5.3	3.93

* The indoor fan was cycled in phase with the compressor for Test QC #11. For all other tests the indoor fan operated continuously.

TABLE 4

Mass (kg) of Liquid and Vapor in Isolated Sections (Calculated from Liquid and Vapor Densities, Isolated Section Volumes and Data of Table 3)

TEST	DESCRIPTION	LIQUID LINE		VAPOR LINE		INDOOR COIL		OUTDOOR COIL		COMPRESSOR & ACCUMULATOR	
		LIQUID	VAPOR	LIQUID	VAPOR	LIQUID	VAPOR	LIQUID	VAPOR	LIQUID	VAPOR
Avg. of QC #7, 8	Cyclic, Off 24 min	.30	.05	.05	.04	2.04	.18	.21	.22	.46	.41
QC#20	Cyclic, On .25 min	.79	.02	.02	.03	.17	.18	1.01	.22	1.18	.30
QC#21	Cyclic, On .50 min	.92	0	.01	.02	.20	.12	1.13	.24	1.13	.20
QC#16	Cyclic, On 1.00 min	1.09	0	0	.01	.10	.08	1.33	.20	.92	.15
QC#17	Cyclic, On 1.50 min	1.18	0	0	.01	.08	.09	1.33	.19	.84	.15
QC#18	Cyclic, On 2.00 min	1.26	0	0	.01	.15	.08	1.30	.19	.78	.15
QC#19	Cyclic, On 4.00 min	1.39	0	0	.01	.19	.09	1.36	.19	.52	.17
Avg. of QC #9, 10	Cyclic, On 6.00 min	1.43	0	.01	.01	.23	.10	1.47	.19	.30	.18
Avg. of QC#3,5,6	Steady-state	1.43	0	0	.02	.31	.11	1.64	.18	.01	.20

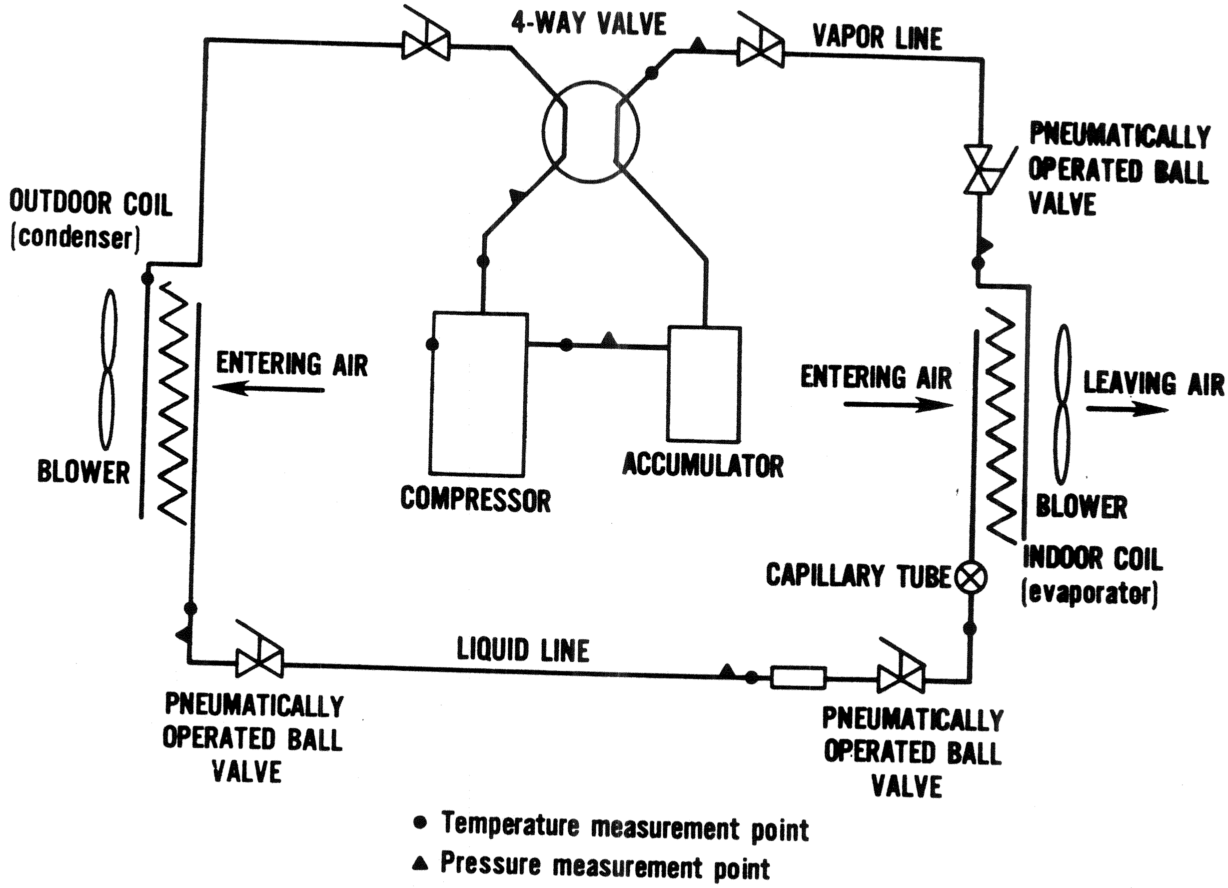

Figure 1. Schematic of NBS heat pump test apparatus

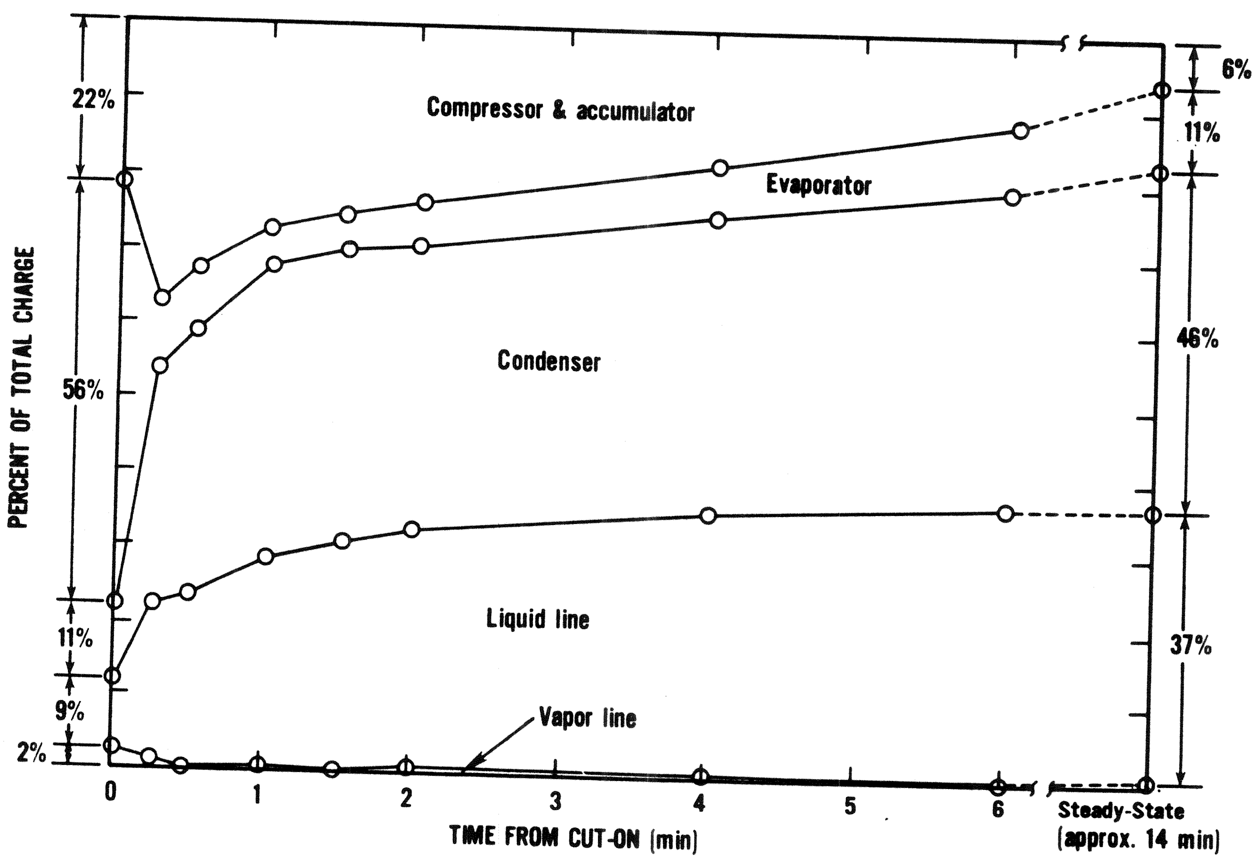

Figure 2. Refrigerant migration during an "on" cycle

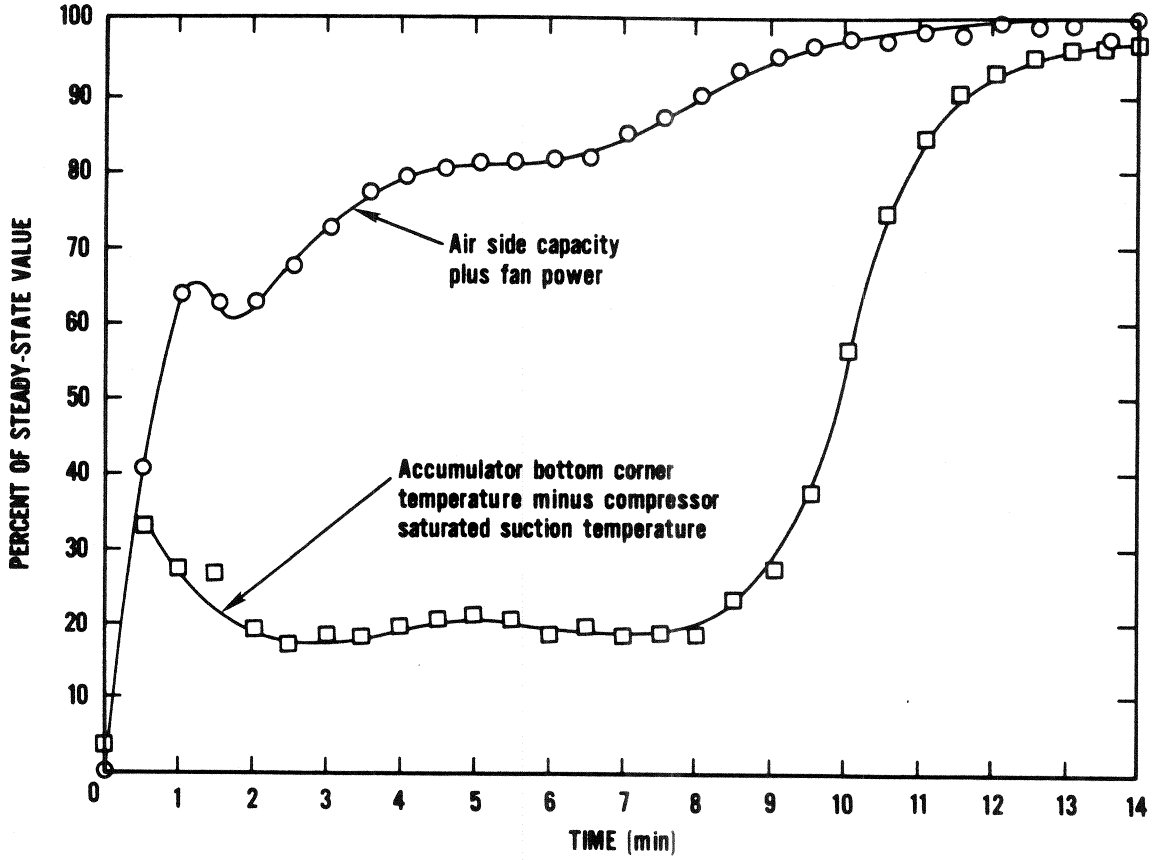

Figure 3. Comparison of time for capacity to reach steady-state to time required to empty accumulator

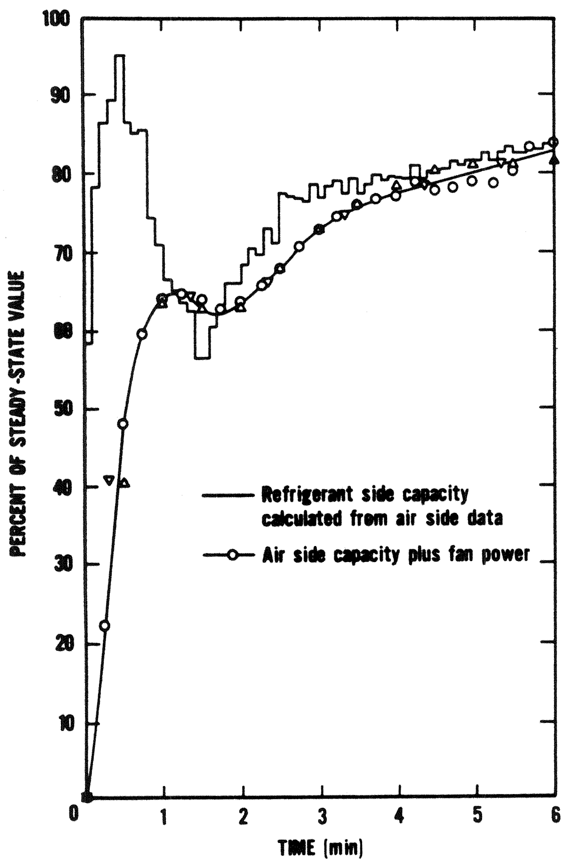

Figure 4. Comparison of transient air side capacity to coincident transient refrigerant side capacity

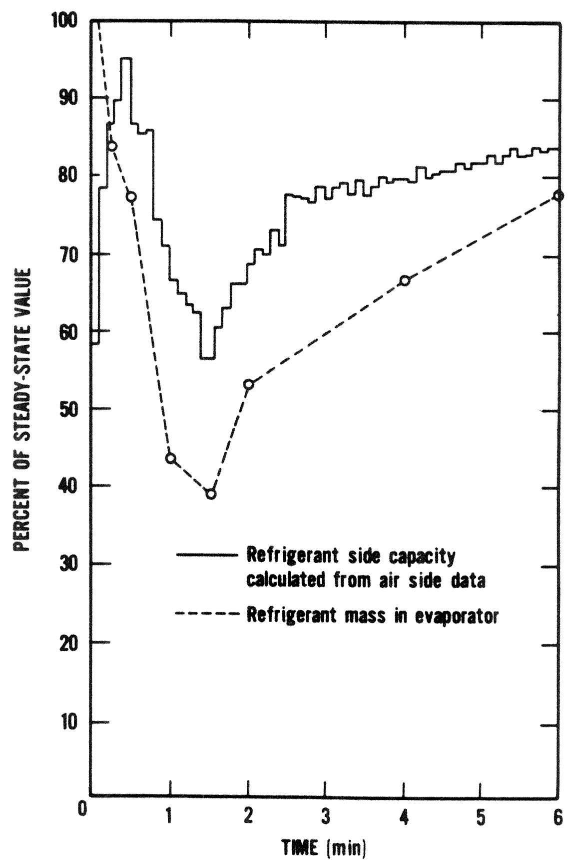

Figure 5. Comparison of refrigerant side capacity to refrigerant mass in evaporator

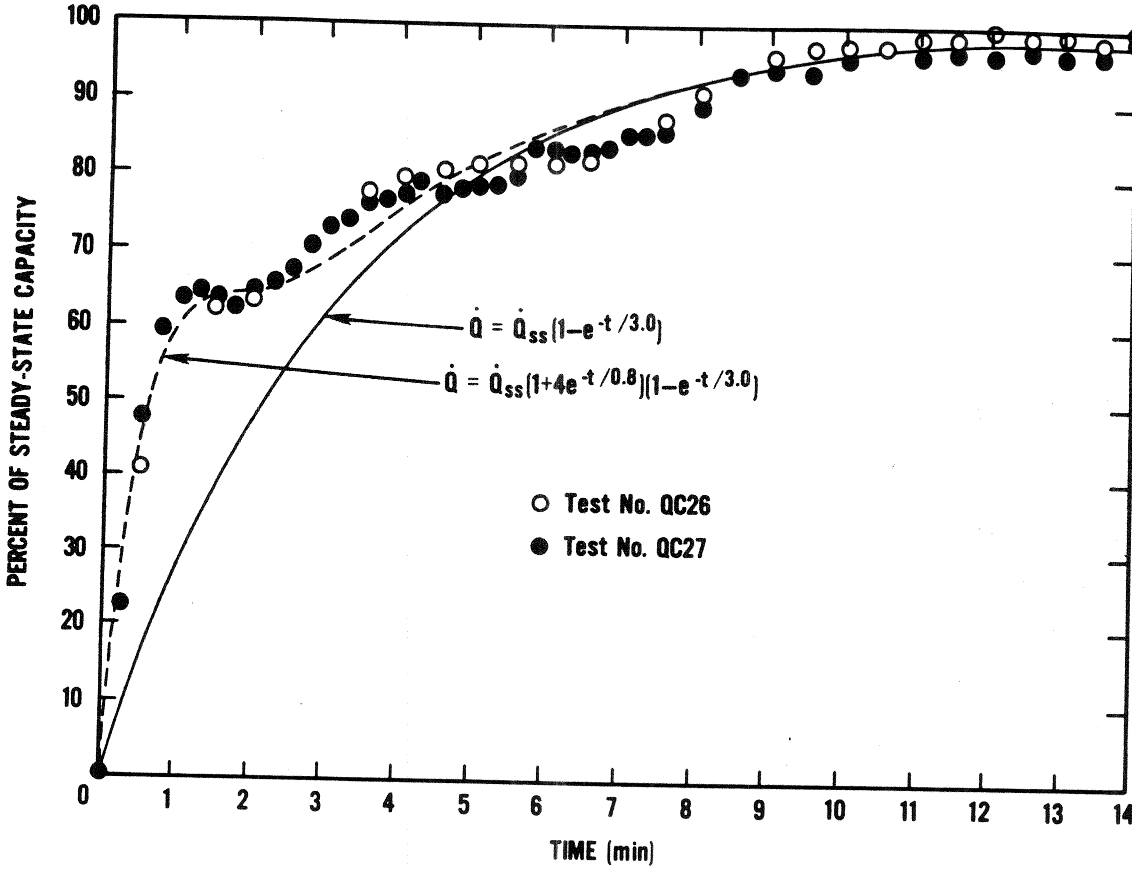

Figure 6. Regressive fit to experimental transient normalized capacity data

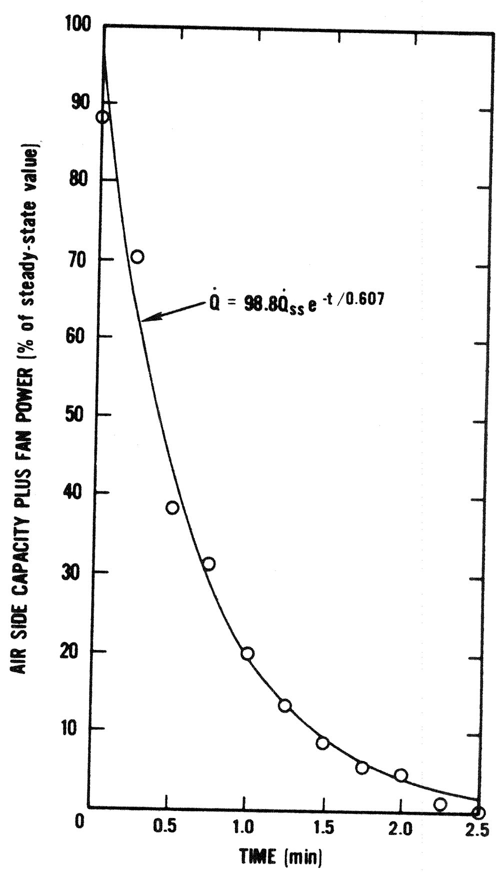

Figure 7. Capacity decay after compressor shut-off with continuous fan operation

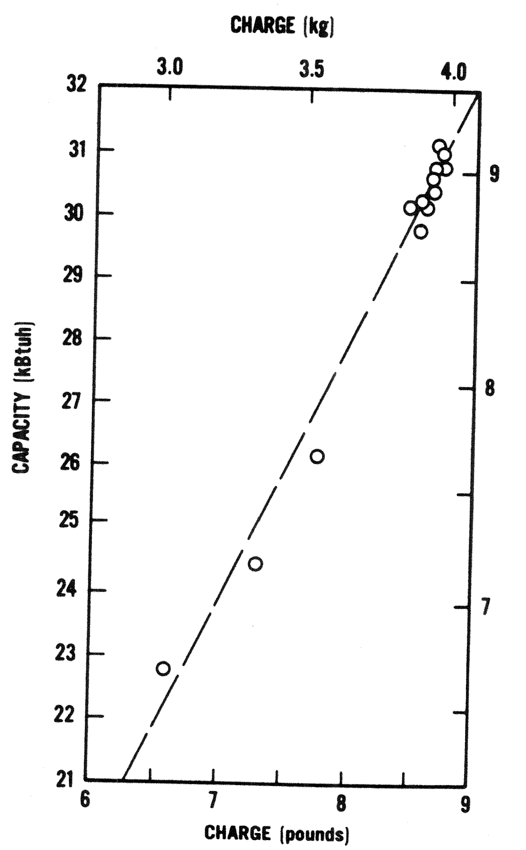

Figure A1. Unit capacity as a function of charge

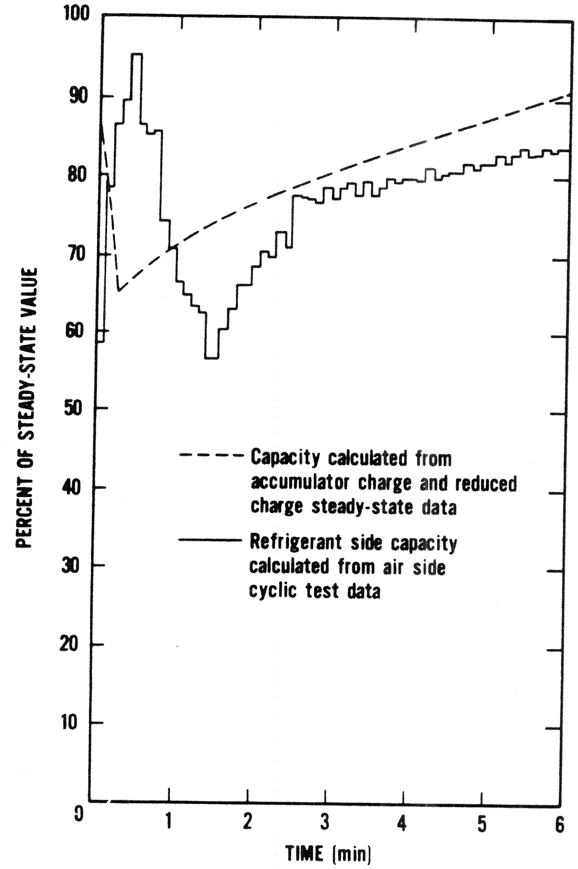

Figure A2. Comparison of refrigerant side capacity to capacity calculated from accumulator content and reduced charge steady-state data

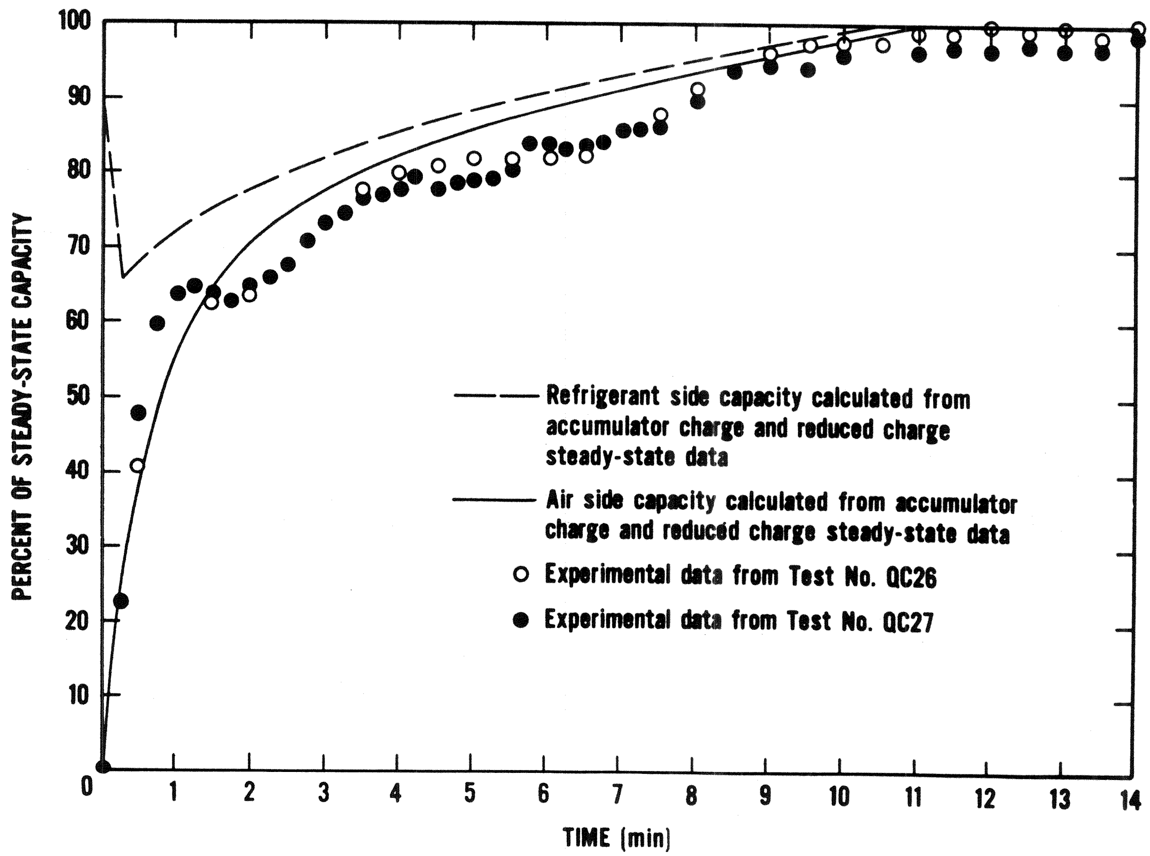

Figure A3. Comparison of air side capacity calculated from accumulator liquid charge and reduced charge steady-state data to experimental air side data

DISCUSSION

O.V. NUSSBAUM, Consulting Engineer, Newtown, PA: Are there any plans to continue this research with thermostatic expansion valves since many split system air conditioners use expansion valves, not capillary tubes?

MULROY: We have no plans for additional migration work. The results in this report suggest that thermostatic expansion valves that do not allow migration during the off cycle or transient flooding on start-up would substantially eliminate that portion of the cyclic degradation which results from the trapping of refrigerant in the accumulator.

J.A. PIETSCH, Dallas, TX: What was the rated cooling capacity of the unit that was tested?

MULROY: The capacity of the unit as charged for these tests was 29,900 Btu/hr at the DOE "A" rating condition. The charging criterion used (15 F superheat leaving the evaporator) would have resulted in a lower capacity and charge than a manufacturer would have been likely to use in rating.

PIETSCH: The volume of the liquid line and the vapor line are approximately the same. In a typical heat pump system with lines of equal length, the vapor line volume is approximately four times the volume of the liquid line. What were the dimensions of the vapor and liquid line in this test?

MULROY: The high liquid line volume was the result of inclusion of a flow measurement board incorporating a turbine flow meter, four hand valves and their associated tubing to allow single direction flow in either operating mode (heating or cooling). A reduction in liquid line length would result in a charge reduction, less refrigerant being trapped in the accumulation on start-up and, therefore, less time to empty the accumulator and a reduction in cyclic degradation.

J. MCCORKEL, Bohn Heat Transfer, Danville, IL: The state of the oil film inside the tubes (i.e. dry, film droplets) will affect capacity. Do you consider its state as it affects capacity? Could it explain the apparent lag between the actual capacity and calculated capacity curves in Figure A2 "Comparison of refrigerant side capacity to capacity calculated from accum- ulator content and reduced charge steady-state data?"

MULROY: Oil film effects were not considered.